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A Combined Thermal System with an Air-cooled Organic Rankine Cycle (ORC)

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Abstract

A combined heat and power (CHP) system with an air-cooled organic Rankine cycle (ORC) is proposed to relieve the shortage of the water resources. In the completely closed thermal system, the steam condenser of the power plant is simultaneously used as the evaporator for the ORC system, and the organic working fluid (OWF), R245ca, from the turbine exhaust is condensed by the ambient air. The performance of the combined thermal system is calculated, and the single and double recuperation are applied to raise the system performance. Furthermore, due to the sensitivity of the air-cooled ORC condenser to the ambient wind, the variational condition performance of the air-cooled ORC condenser at different ambient temperatures is calculated.

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Keywords: water resource; CHP system; air-cooled; ORC; ambient temperature

1. Introduction

Thermal power plants are extensively developed in the north of China, and the corresponding turbine exhaust or extraction steam is applied to provide heat to the industrial enterprises or consumers. As a result, back pressure steam turbines are generally equipped in the combined heat and power (CHP) system. However, for the thermal power plant, the power load is determined by the heat load [1].

Organic Rankine cycle is an excellent scheme to utilize the low grade heat such as the waste heat from industry or geothermal energy recent years [2-4]. Liu [5] presents a cogeneration system in which a back pressure steam turbine generating unit is coupled with an organic Rankine cycle. R113 is selected as the working fluid in the ORC cycle, and the performance of the ORC system is investigated at the rating condition of the steam cycle.

However, the CHP system with an air-cooled ORC condenser is not involved in the previous literature, and the coupling effect between the steam cycle and the ORC cycle is presented. It is assumed that there is no heat load in the steam system, and all the steam turbine exhaust is utilized to heat the

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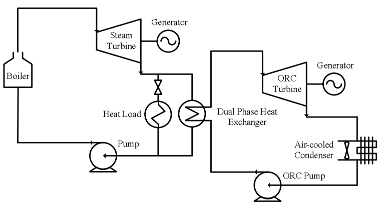
organic working fluid (OWF), R245ca, in the dual phase heat exchanger. Furthermore, the variational condition characteristics of the ACC with the ambient temperature are also calculated.

Nomenclature

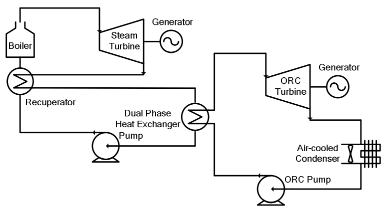
h	heat transfer coefficient, w/(K·m^2)	*m*	mass flow rate, kg/s
p	pressure, MPa	T	temperature, K
TD	temperature difference, K	*W*	output power of the turbine, kW
Q	heat duty of the boiler, kW	*η*	efficiency
Subscripts			
a	air	*bp*	back pressure
c	consumption	*p*	pump
r	recuperation	*ref*	reference
s	steam	*t*	total, turbine

2. CHP system description

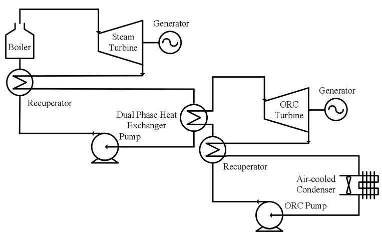
The CHP power plant with steam and organic working medium are presented in Fig. 1. It is assumed that there is no heat load during the investigation, and the heat load exchanger is not shown in Fig. 1 (b) and (c). It is seen that two thermal cycles are connected through the dual phase heat exchanger (DPHE), and the turbine exhaust is applied to heat the organic working medium, which evaporates to drive the turbine in the ORC system. Furthermore, ambient air is applied to condense the waste heat from the ORC turbine to save the scarce water resources.



(a) Simple CHP system



(b) CHP system with steam recuperation



(c) CHP system with steam and ORC recuperation

Fig.1 Layout for three types of CHP systems

3. Analysis and discussion

The current investigation aims to the comprehensive performance of the CHP system as well as the correlation among the components of the thermal system. Before the analysis, it is necessary to make some assumptions to simulate the performance of the thermal cycle, and the relevant parameters are listed in Table. 1.

Table 1 Assumed parameters for the CHP system

Parameters in the CHP thermal system	value
p_s, MPa	1.27
T_s, K	613
m_s, kg/s	1.88
TD_r, K	4
TD_{DPHE}, K	5
TD_{ORCr}, K	8
TD_{ACC}, K	15
T_a, K	306
η_t, %	0.8
η_p, %	0.85
m_{aref}, kg/s	537.5
h_{ref}, w/(k·m^2)	33

3.1. Simple CHP system

The performance of the CHP system based on the assumed parameters is shown in Table 2 when the back pressure of the steam turbine varies from 0.2Mpa to 0.4Mpa. It is seen that the total power of the CHP system drops from 1104.3kW to 984.7kW while the power consumption increases from 90.1kW to 99.7kW. Taking the boiler heat duty into consideration, the alternation of the back pressure finally results in the reduction of the thermal efficiency.

Table 2 Calculation results of the CHP system performance

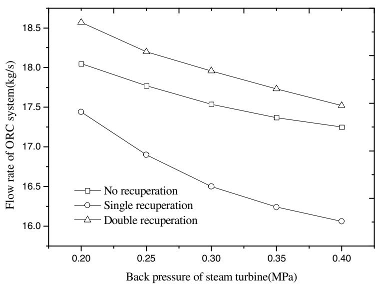
p_{bp}, Mpa	0.2	0.25	0.3	0.35	0.4
p_{ORC}, Mpa	1.2	1.4	1.5	1.7	1.9
W_t, kW	1104.3	1058.4	1021.2	1002.8	984.7
W_c, kW	90.1	92.9	94.0	97.0	99.7
Q, kW	4958.0	4903.4	4856.0	4816.3	4780.0
η_{CHP}, %	20.5	19.7	19.1	18.8	18.5

3.2. Effect from the recuperation

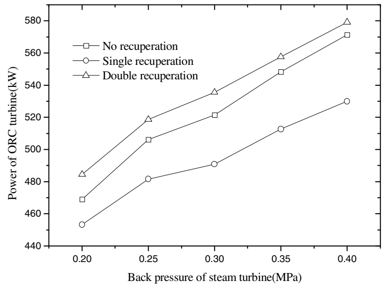
The performance of the ORC system with the back pressure of the steam turbine is presented in Fig. 2 for the three designated systems. It is found that the double recuperation in the CHP system contributes a maximum mass flow rate of the ORC system, while the single recuperation in the steam system results in a minimum flow rate.

Corresponding to the trend of the mass flow rate described, the double recuperation system has the highest turbine power of the ORC system. Obviously, in order to keep the fixed heat transfer temperature difference of the dual phase heat exchanger, the OWF pressure into the turbine also rises, and the enthalpy drop in the ORC turbine is raised significantly. Finally, the turbine power rises with the increase of the steam turbine back pressure although the trend of the mass flow rate is reversed.

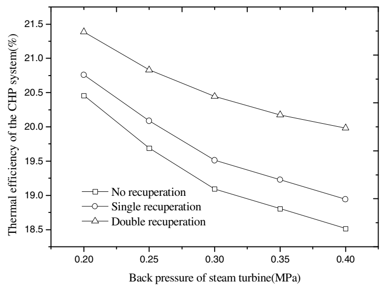
It is also seen that both the single and double recuperation are benefit to raise the comprehensive performance of the combined cycle. Compared to the simple CHP system, the absorbed heat in the boiler is also reduced because of the heated inflow after the recuperation, and finally the single recuperation system is superior to the simple CHP system.



(a) Mass flow rate



(b) ORC turbine power



(c) Thermal efficiency

Fig.2 Performance of the ORC system at different back pressures of the steam turbine

3.3. Effect from the ambient temperature on the ACC performance

The influence from the ambient temperature is investigated based on the double recuperation system at the back pressure of 0.3Mpa for the steam turbine, and the relevant design parameters at the ambient temperature of 306K are listed in Table 3.

Table 3 ACC design parameters at ambient temperature of 306K

Heat duty (kW)	Heat transfer area (m^2)	Fan flow rate (m^3/s)	LMTD (K)	Heat transfer coefficient ($w/(K \cdot m^2)$)
3567.4	9138.7	537.5	11.8	33

The air-cooled condenser will respond to the alternation of the ambient temperature, and the fan flow rate as well as the power of the forced draught fan will change. Furthermore, the heat transfer coefficient also differs with the mass flow rate, and the corresponding relation can be illustrated as [6]:

$$h = h_{ref}(m_a / m_{aref})^{0.283} \tag{1}$$

where h is the total heat transfer coefficient, $w/(k \cdot m^2)$, h_{ref} the reference or design coefficient, m_{aref} the design mass flow rate of air through the ACC, 537.5kg/s.

The calculated variational condition characteristics for the air-cooled condenser at different ambient temperatures are presented in Fig. 3, in which the equivalent heat transfer area implies the necessary area to satisfy the heat duty of the ACC at the designed mass flow rate, 537.5kg/s. It is found that the mass flow rate rises from 416.2kg/s at 304K to 733.4kg/s at 308K, and an alternation range of 59% based on the design flow rate is obtained. As a result, the axial fan mode must be adjusted to generate the necessary flow rate, and the fan consumption power will be changed to accommodate the resistance of the ACC. Simultaneously, the equivalent area is also calculated, and it differs from

7793.5m^2 to 11066.2 m^2 with a range of 35.8%. It is inferred that the design ambient temperature must be rational to have an optimal heat transfer area, and then an excellent, comprehensive performance of the ACC can be obtained in the range of the ambient temperature in a year.

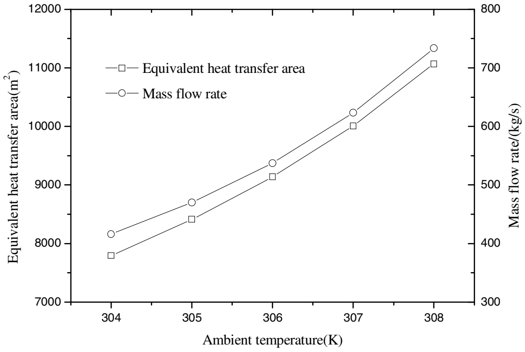


Fig.3 Variational condition characteristics of the air-cooled condenser at different ambient temperatures

4. Conclusions

The results in the paper are significant to solve the coordination problem between the heat load and electrical load for the general thermal power plants. The waste heat from the back pressure turbine is well recovered by the ORC system. The ORC turbine has a comparative power compared to the steam turbine at the assumed thermal parameters, and the ORC turbine power varies with the back pressure of the steam turbine. An increase amplitude of 1.35% for the thermal efficiency is achieved through the application of double recuperation in the CHP system. Simultaneously, the equivalent area differs from 7793.5m^2 to 11066.2 m^2 with a range of 35.8% when the ambient temperature increases from 304K to 308K. The fan mode must be alternated to accommodate the change of the ambient temperature.

Acknowledgements

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